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Development of a more robust correlation for predicting heat transfer performance in oscillatory baffled reactors

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Abstract

The oscillatory baffled reactor has been well-characterised in most areas of flow reactor performance (mixing, mass transfer, multi-phase operation *etc*), with the exception of heat transfer, where comparatively few data exist in the literature. Here, a robust investigation of heat transfer in the "standard", 26mm diameter, oscillatory baffled reactor is presented which goes beyond the parametric limitations of previous studies.

5-fold Nusselt number increases over steady unbaffled flows are shown to be achievable. The maximum enhancement attributed to the oscillatory flow component alone compared to the steady-flow, baffled case is shown to be 1.7-fold. The degree of heat transfer enhancement is shown to plateau when the oscillatory flow Reynolds number exceeds 1300, indicating that a radial mixing limit has been reached.

A new correlation for predicting heat transfer coefficients in oscillatory baffled reactors has been developed. Based on the data generated here, it is accurate to +/- 30% across the experimental range of the study. The correlation has been further validated using literature data, and shown to be the most robust correlation to date for predicting heat transfer performance in oscillatory baffled reactors.

Key Words

Oscillatory baffled reactor; heat transfer enhancement; process intensification

1. Introduction

The oscillatory baffled reactor (OBR) is a continuous tubular flow reactor in which an oscillatory flow is imposed onto a relatively small net flow within a baffled tube. This results in the formation and dissipation of vortices either side of the baffles throughout each flow reversal cycle, effectively creating a series of well-mixed "tanks-in-series". This leads to plug flow behaviour even though net flow conditions are laminar. This allows the rate of mixing, heat and mass transfer to be decoupled from the net flow rate, which in turn largely decouples length - turbulence - velocity design.

Oscillatory baffled flows are characterised using four dimensionless groups:

- The net flow Reynolds number, Re_n, which defines the net flow condition.
- The oscillatory Reynolds number, Re_o, which gives a measure of the oscillatory flow intensity.
- The velocity ratio, Ψ. The ratio of the oscillatory flow to net flow intensity.
- The Strouhal number, Sr. A measure of eddy propagation in the pipe.

$$Re_n = \frac{\rho u D}{\mu}$$

[Eq. 1]

$$Re_o = \frac{x_0 \omega \rho D}{\mu}$$

[Eq. 2]

$$\Psi = \frac{Re_o}{Re_n}$$

[Eq. 3]

$$Sr = \frac{D}{4\pi x_0}$$

[Eq. 4]

Here ρ is the fluid density (kg/m^3), u the superficial net flow velocity (m/s), D the pipe diameter (m), μ the fluid viscosity (kg/ms), x_0 is the centre-to-peak amplitude of oscillation and ω the angular frequency of the oscillation cycle (ω = 2πf, where f is the frequency of oscillation in Hz).

Mixing [1, 2, 3, 4] and mass transfer [5, 6] in OBRs have been well defined in previous studies, however there has been relatively little research into heat transfer. Most recently, Solano et al [7] studied heat transfer enhancement in helically baffled meso-scale OBRs as part of a CFD study into the flow structures generated in this type of OBR. Their results showed a Nusselt number enhancement of up to 4-fold compared to steady, unbaffled flow, which was well explained by the velocity streamlines and vectors from the flow-pattern analysis section of the paper. However, no experimental validation was presented and no correlation was proposed for predicting OBR Nusselt numbers.

Mackley *et al* [8] presented the results of a preliminary study on heat transfer in pulsatile and oscillatory baffled flows. The preliminary data obtained in this study showed that a Nusselt number enhancement of up to 6-fold could be achieved using oscillatory baffled flows compared to steady flow in smooth tubes. Mackley and Stonestreet [9] then expanded on the findings of the previous paper in a further experimental study over a wider range of oscillation frequencies and amplitudes. They showed that up to a tenfold increase in the Nusselt number is possible using oscillatory baffled flows compared to steady, unbaffled tubes. A Nusselt number correlation for oscillatory baffled flows was devised (Eq. 5). While the authors acknowledged that this was a purely phenomological model based only on their data set, they did also state that it shows the correct behaviour, thereby allowing it to be used over an extended range of Re_n and Re_o. The inclusion of the Prandtl number (Pr) suggests that the correlation should be valid for all liquids, although only one fluid was evaluated in the study, an engine oil of average Pr = 73. As a result, this correlation is accepted as the best estimate to predict heat transfer coefficients in OBRs [10].

$$Nu = 0.0035 {Re_n}^{1.3} Pr^{\frac{1}{3}} + 0.3 \left[\frac{{Re_o}^{2.2}}{(Re_n + 800)^{1.25}} \right]$$

[Eq. 5]

In Eq. 5 the first term corresponds to the steady-flow contribution to heat transfer and is similar to the Dittus-Boelter correlation for heat transfer in turbulent flows. The second term accounts for the heat transfer enhancement observed due to the oscillatory flow component. The correlation suggests that the enhancement observed wanes as Re_n becomes larger than Re_o meaning that the Nusselt number tends towards that for steady-baffled flow at higher net flow rates. For constant net flow Reynolds number, the Nusselt number will increase according to an approximately squared relationship for increase in the oscillatory flow Reynolds number.

It appears that the correlation is not valid for liquids of relatively low Pr. For example, if the correlation is used to predict the Nusselt number of an OBR using water as the working fluid, the results displayed in Fig. 1 are obtained (mean fluid temperature of 40°C, mean Pr of 4.43).

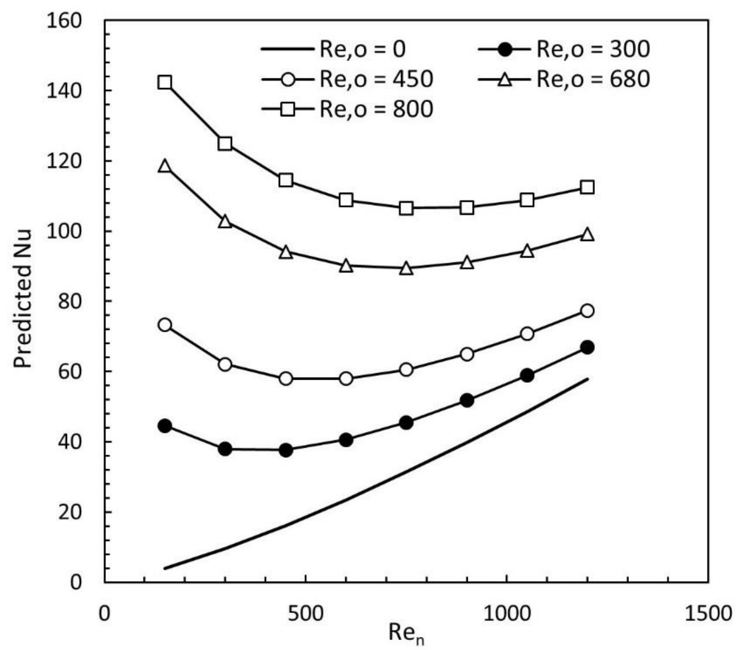

Figure 1. Plot of predicted Nusselt number vs Re_n and Re_o for water

Fig. 1 shows that the correlation predicts that, for constant Re_o, minima exist as Re_n increases. This behaviour was not observed in the study by Mackley and Stonestreet [9] and there are no reports of it occurring. The explanation for this result lies in the form of the correlation equation and the relative size of the two terms. In Fig. 1, the second term in the correlation reduces in size as the net flow Reynolds number increases by an amount which cannot be sustained by the increase in the first term, thereby leading to a minimum in the function. This behaviour is masked at higher values of Pr (as in the original study, Pr = 73), as in all cases the value of the first term is significantly larger than the second term. This result suggests that the correlation may not be valid for lower Pr liquids and that further investigation is required, to find a more general expression.

It is also necessary to perform heat transfer experiments over a wider range of oscillatory flow intensities. In Mackley and Stonestreet [9] the highest oscillatory Reynolds number tested was 800. Hence, for Re_n > 800 (around 40% of the dataset) the velocity ratio was less than 1. Under these conditions, full flow reversal would not be achieved and is not representative of general OBR design (velocity ratios of greater than 2 are typically used to maintain the compact design of the OBR). It is therefore possible that the hypothesis that oscillatory flow has little effect on heat transfer at higher net flows and that the heat transfer coefficients tend towards those of steady baffled flows may only be due to the fact that the oscillation intensity investigated was too weak.

The aim of this paper is to increase understanding of heat transfer phenomena in OBRs and to generate a new, robust correlation for predicting the Nusselt number by exploring a greater parametric range than previously studied.

2. Experimental

2.1 Materials, Apparatus and Methods

A laboratory-scale countercurrent annular tube heat exchanger was used to conduct the experiments, as shown in Fig. 2 (schematic diagram) and Fig. 3 (photograph of rig). The inside tube (OBR-side) was a 26.2mm i.d. copper tube with a wall thickness of 0.9mm, while the outer-tube (shell-side) was a 39.6mm i.d. copper tube. The active length of the heat exchanger was 500mm. Orifice baffles were used on the OBR-side (13mm orifice; 52mm baffle spacing).

The cold fluid flowed on the shell-side of the heat exchanger and was provided by mains water at a constant 3 l/min flowrate. The cooling flow was at a temperature of 12-15^{o}C throughout the experimental programme meaning that variation in thermophysical properties were negligible.

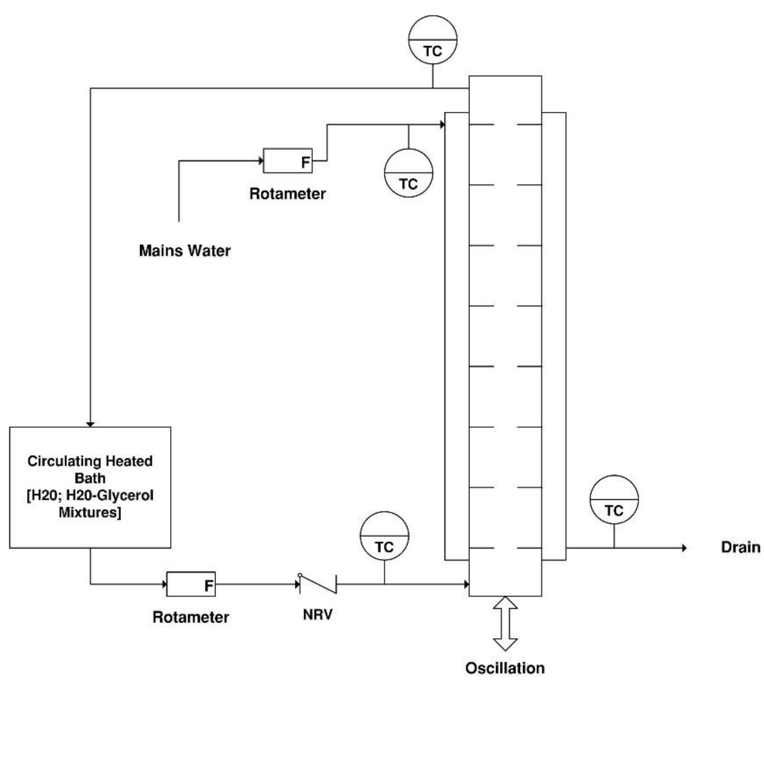

Figure 2. Schematic of rig

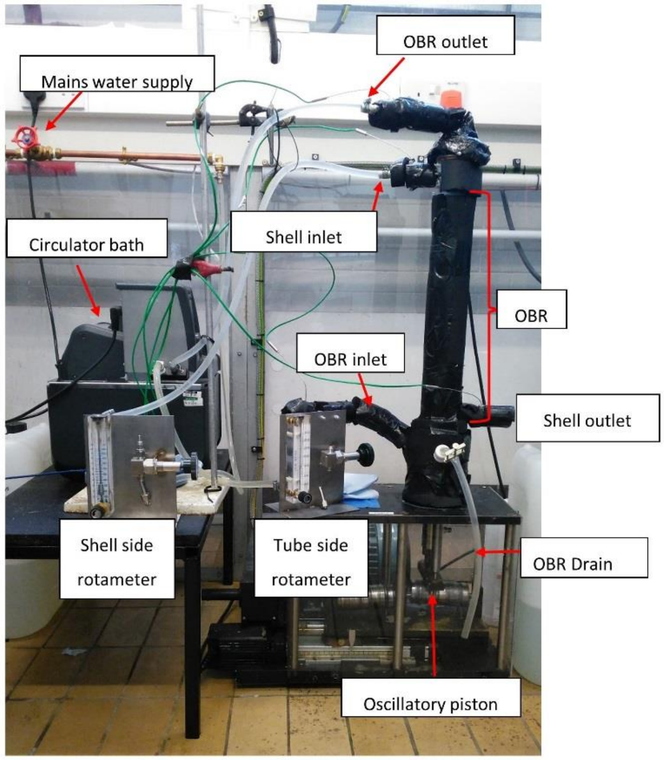

Figure 3. Annotated photograph of experimental rig

Two OBR-side fluids were used: deionised (DI) water and 25wt% glycerol-DI water solution which allowed a two-fold variation in Pr, as highlighted by the selected thermophysical data shown in Table 1. A Grant Circulating bath (12l, TFX200 series) was used to both maintain the hot-side inlet temperature (50°C) and to provide the net flow to the OBR. Oscillations were provided to the OBR by a piston at the inlet which was operated by a connecting rod and yoke arrangement. This was driven by an electric variable-speed drive, allowing the frequency to be adjusted. The amplitude of oscillation was varied using a lateral motor which adjusted the connecting rod pivot point, allowing the amount of displacement caused by the piston to be adjusted.

Table 1. Selected thermophysical data for the two fluids used in this study (at 40°C, average tube-side temperature throughout the experimental programme)

	Specific Heat Capacity (J/kg.K)	Thermal Conductivity (W/m.K)	Viscosity (10^{-3} Pa.s)	Density (kg/m^3)	Pr
DI Water	4180	0.623	0.655	992	4.40
25wt% Glycerol-Water	3740	0.523	1.24	1063	8.90

Fluid temperatures were measured at the heat exchanger terminals using four k-type, 1.5mm diameter thermocouples which were calibrated according to BS1041-4. A Pico Log TC-08 and Windows 7 PC were used to record the thermocouple data. Omega FL-2051 and FL-2069 rotameters were used to set the OBR net flow and shell-side flow rates respectively.

The heat exchanger was heavily lagged to ensure that a heat balance between the hot and cold fluids was maintained throughout the experimental programme.

The OBR-side net flow rate and oscillatory flow condition were varied across the entire range of the experimental equipment according to the parameter range listed in Table 2. The effect of the Strouhal number was not investigated in this study as it has been extensively studied previously [7, 9] and shown to have no significant impact on heat transfer rates for constant oscillatory Reynolds number.

Table 2. Range of flow parameters tested in study

Range of net flow rates (ml/s)	5.00 - 18.3
Range of centre-to-peak oscillation amplitude (mm)	0.00 - 6.00
Range of oscillation frequency (Hz)	0.00 - 5.84
Range of velocity ratio (-)	0.00 - 10.1

All readings were taken once steady states had been established. All runs were performed in triplicate.

2.2 Treatment of Results

The film heat transfer coefficient of the OBR, h_{OBR}, was found using the heat transfer resistances in series model as follows:

$$R_{tot} = \sum R_i = R_{OBR} + R_{wall} + R_{shell} = \frac{1}{U}$$

[Eq. 6]

Here R_{tot} is the total heat transfer resistance (Km^2/W), R_{OBR} is the OBR-side resistance (Km^2/W), R_{wall} is the conductive resistance through the heat exchanger wall (Km^2/W), R_{shell} is the shell-side resistance (Km^2/W) and U is the overall heat transfer coefficient (W/m^2K). The individual resistances are defined as follows for a thin-walled tube-in-tube heat exchanger (i.e. heat transfer area on both sides is approximately equal):

$$R_{OBR} = \frac{1}{h_{OBR}}$$

[Eq. 7]

$$R_{shell} = \frac{1}{h_{shell}}$$

[Eq. 8]

$$R_{wall} = \frac{x}{k_{wall}}$$

[Eq. 9]

Here h_{OBR} and h_{shell} are the OBR-side and shell-side film heat transfer coefficients (W/m^{2}K), x is the wall thickness (m) and k_{wall} is the thermal conductivity of the wall (W/mK).

In this case the wall resistance was ignored as it was negligible within the accuracy of the experimental procedure. Typical overall heat transfer coefficients, U, obtained during experiments were in the region of 800-1200W/m^{2}K. Hence, the total resistance of the circuit was in the region of 0.000833-0.00125 Km^2/W. The thermal conductivity of copper is approx. 400W/mK within the experimental range of temperatures, and the wall thickness was 0.0009m. The wall resistance is therefore 2.25 x 10^{-6} Km^2/W which is approximately 0.3% of the minimum value of the total heat transfer resistance.

The overall heat transfer coefficient, U, is found according to the heat transfer duty as follows:

$$Q = UA\Delta T_{LM}F = Q_{OBR}$$

[Eq. 10]

Here Q is the overall heat transfer duty (W), A is the heat transfer area (m^2), ΔT_{LM} the log mean temperature difference (K) (Eq. 7), F is the correction factor which is 1 for pure counter-current flow, and Q_{OBR} is the duty (W) as calculated using data for the OBR-side (Eq. 6), which is equal to the duty as calculated using data for the shell-side in accordance with the system being in heat balance.

$$Q_{OBR} = \dot{m}_{OBR} c_p \Delta T_{OBR}$$

[Eq. 11]

Here $\dot{m}_{OBR}$ is the mass flow rate (kg/s) of the OBR-side fluid, c_p is the specific heat capacity of the OBR-side fluid (J/kgK) and ΔT_{OBR} is the temperature difference (K) between the OBR inlet and outlet.

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} = \frac{(T_{OBR,out} - T_{shell,in}) - (T_{OBR,in} - T_{shell,out})}{\ln \frac{(T_{OBR,out} - T_{shell,in})}{(T_{OBR,in} - T_{shell,out})}}$$

[Eq. 12]

Here $T_{OBR,in}$ and $T_{OBR,out}$ are the inlet and outlet temperatures of the OBR-side (K), and $T_{shell,in}$ and $T_{shell,out}$ are the inlet and outlet temperatures of the shell-side (K).

The shell-side heat transfer coefficient was calculated using a Wilson-plot [11]:

1. Base-line experiments were carried out with no baffles and a steady flow over the entire range of net flow Reynolds numbers on the OBR-side whilst keeping the shell-side condition constant (Re_{shell} = 750, Pr_{shell} = 8.5).
2. The OBR-side heat transfer coefficient was assumed to be proportional to Re_n^m, where m was determine iteratively.
3. 1/U vs 1/Re_n^m was plotted. The extrapolated trend line at the intercept Re_n^m=0 corresponds to the point of infinitely small tube-side resistance. Hence, this value is taken as the shell-side resistance, 1/h_{shell}, as the wall resistance can be ignored (see above). This was found as 4.61 x 10^{-4}, h_{shell} = 2170W/m^{2}K. This is assumed constant throughout all experimental runs as (i) the shell-side temperature does not significantly deviate between experiments (inlet temperature was approximately constant and the shell-side flow is at a large excess, meaning the temperature rise

was relatively low and the difference negligible between runs), hence the thermophysical properties of the shell-side fluid will be approximately constant throughout the experimental programme and (ii) the shell-side volumetric flow rate, and therefore velocity, are kept constant. Furthermore, this value was shown to be repeatable when the test was repeated for each of the OBR-side fluids.

The resistances in series model reduces to the following, where 1/U was determined experimentally according to Eq. 10 and Eq. 11, and h_{OBR} was found by simple rearrangement:

$$\frac{1}{U} = \frac{A.\Delta T_{LM}}{(mc_p\Delta T)_{OBR}} = \frac{1}{h_{OBR}} + \frac{1}{h_{shell}}$$

[Eq. 13]

$$h_{OBR} = \frac{1}{\frac{A.\Delta T_{LM}}{(mc_p\Delta T)_{OBR}} - \frac{1}{h_{shell}}} = \frac{1}{\frac{A.\Delta T_{LM}}{(mc_p\Delta T)_{OBR}} - \frac{1}{2170}}$$

[Eq. 14]

This is typically displayed as the Nusselt Number, the dimensionless ratio of convective to conductive heat transfer:

$$Nu = \frac{h_{OBR}D}{k}$$

[Eq. 15]

Here D is the OBR tube diameter (0.0262 m), and k is the OBR fluid thermal conductivity (W/mK).

3. Results and Discussion

3.1 Effect of baffle

Initial experiments were conducted without fluid oscillations to establish the base-line heat transfer coefficients for the system, with and without baffles, as shown in Fig. 4.

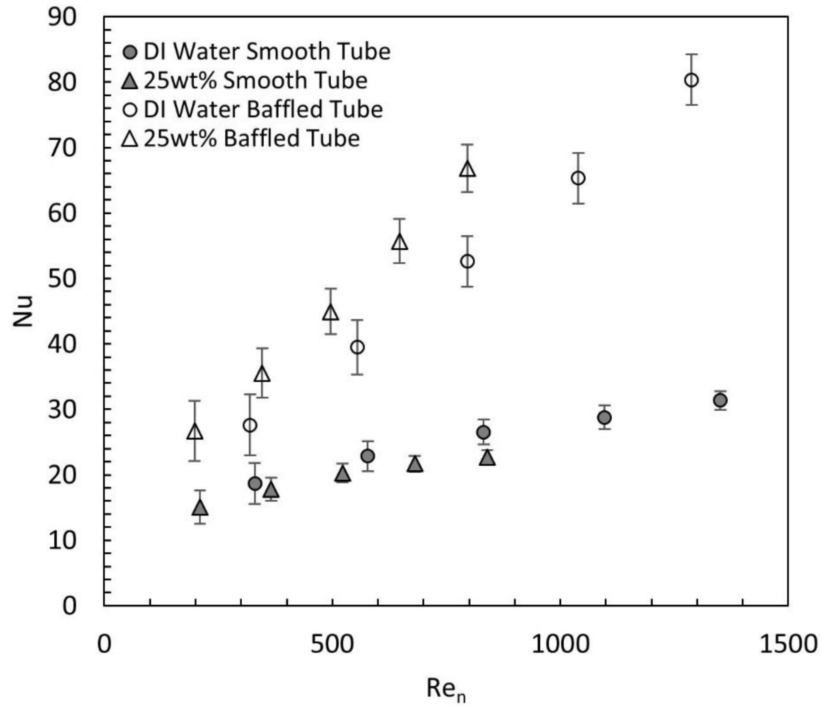

Fig. 4. Nusselt numbers observed for steady flow conditions (without oscillations)

Fig. 4 shows that for each fluid, a significant enhancement in the heat transfer coefficient occurs when baffles are added, of up to 3-fold is observed at the highest flow rates, and up to 1.5-fold at the lowest flow rates. The widening of the heat transfer enhancement with increasing Reynolds number is probably due to the flow exhibiting turbulent characteristics upon addition of the baffle, meaning that the Nusselt number is a higher order function of the Reynolds number than the smooth tube, fully laminar case. More generally, the Nusselt numbers for the 25wt% mixture are around 25% higher than those for water (for the same net flow Reynolds number). This is consistent with typical correlations for predicting Nusselt numbers in tubular heat exchangers where the Nusselt number is normally a function of $Pr^{0.33}$ (and also Re_n to some exponent, depending on the flow regime): here the difference in Pr between the two fluids is roughly a factor of 2, and $2^{0.33}$ is approximately 1.25.

3.2 Effect of oscillatory Reynolds number

The effect of oscillatory flow is summarised in Figs 5 (for DI water) and 6 (for 25wt% glycerol mixture). Note: in Fig. 5 and 6 the “baseline” data refers to the steady, unbaffled case.

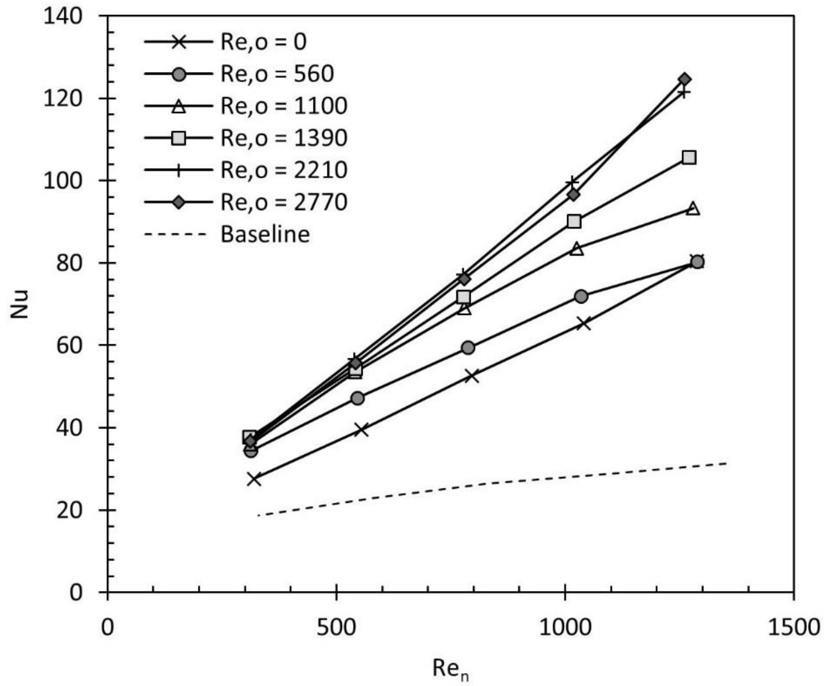

Figure 5. OBR Nusselt Number as a function of the net flow and oscillatory flow Reynolds numbers, DI-water working fluid

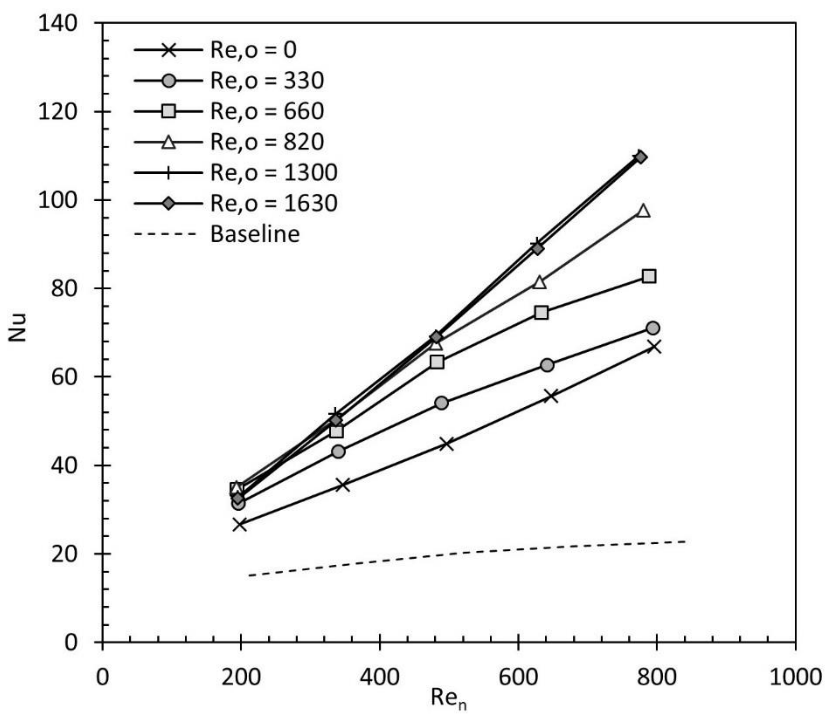

Figure 6. OBR Nusselt Number as a function of the net flow and oscillatory flow Reynolds numbers, 25wt% glycerol working fluid

Fig. 5-6 show that the addition of an oscillatory flow component significantly enhances the Nusselt number. Compared to the steady, unbaffled base-case, enhancement of approximately 2-fold is observed at low net flow rates, and approximately 5.5-fold at high net flow rates. It is also observed for all cases that at higher oscillatory flow Reynolds numbers, the Nusselt numbers begin to converge suggesting that a constant maximum is reached at a specific oscillatory flow Reynolds number. This is further investigated in Fig. 7-8.

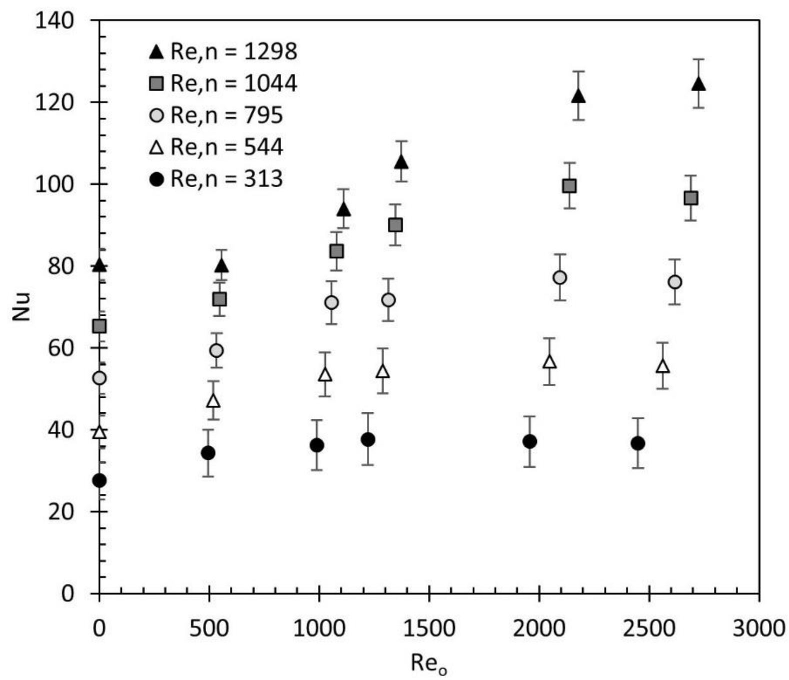

Figure 7. OBR-side Nusselt number as a function of oscillatory Reynolds number for various net flow Reynolds numbers, DI Water working fluid.

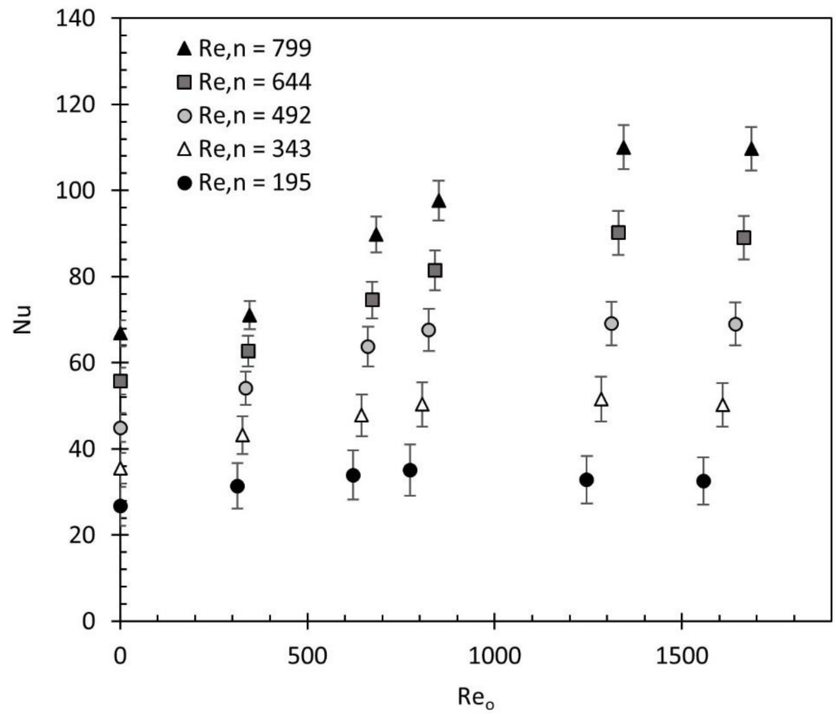

Figure 8. OBR-side Nusselt number as a function of oscillatory Reynolds number for various net flow Reynolds numbers, 25wt% glycerol working fluid

Fig 7-8 show that for both fluids, the Nusselt number rises with increasing oscillatory flow Reynolds number until a maximum is reached at a Re_o of approx. 1300. At this point, the Nusselt number levels off within the experimental range.

This behaviour is analogous to the results of previous mixing studies. Smith and Mackley [12] studied axial dispersion in OBRs over a wide range of diameters, Re_n and Re_o. They found that a minimum exists in axial dispersion at a Re_o of 800-1000. This is theorised to be the point of peak radial mixing within the OBR (presumably limited by the wall radius), and increasing the Re_o beyond this value serves only to increase axial mixing. In heat transfer, the point of peak radial mixing would also be the point of maximum thermal boundary layer disturbance. Hence, increasing the oscillation intensity beyond this point does not further enhance the heat transfer coefficient. In contrast to the axial dispersion study, the enhancement in heat transfer does not appear to wane beyond this value of Re_o (see Fig. 9). This is probably because axial dispersion will have a negligible impact on the local bulk flow temperature (and therefore the temperature difference between the two fluids in the heat exchanger) compared to the effect of the oscillating flow.

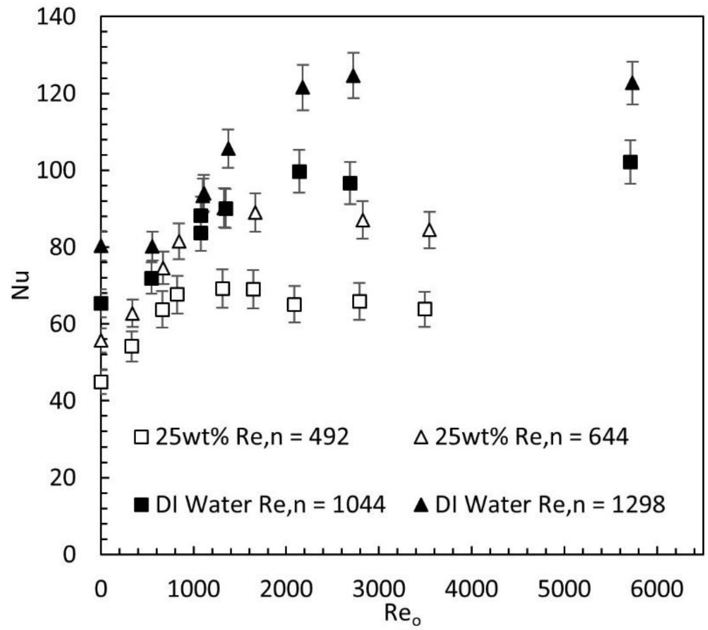

Figure 9. OBR-side Nusselt number over an extended range of oscillatory Reynolds number

3.3 Correlation for predicting heat transfer coefficients in OBRs

The correlation below has been devised using a least-squares curve fitting code based on the experimental data collected during this study. The correlation is analogous to the Dittus-Boelter correlation for turbulent flows with an additional term, Re_o, which is unique to OBRs. No term is included to account for the difference between the bulk flow and wall fluid viscosity as this could not be determined within the experimental methodology. Furthermore, this is unlikely to be significant in an OBR due to the high mixing rates achieved.

For $0 < Re_o \leq 1300$:

$$Nu = 0.022\ Re_n^{0.7} Pr^{0.3} {Re_o}^{0.44}$$

For $Re_o > 1300$

$$Nu = 0.52\ Re_n^{0.7} Pr^{0.3}$$

[Eq. 16]

The accuracy of the correlation is quantified in Fig. 10, where the Nusselt number calculated using the correlation is plotted against the experimentally determined Nusselt number. The results show that all of the data fall within 30% confidence limits, thereby confirming the validity of this correlation for predicting the heat transfer performance of OBRs within the range of experiments investigated here.

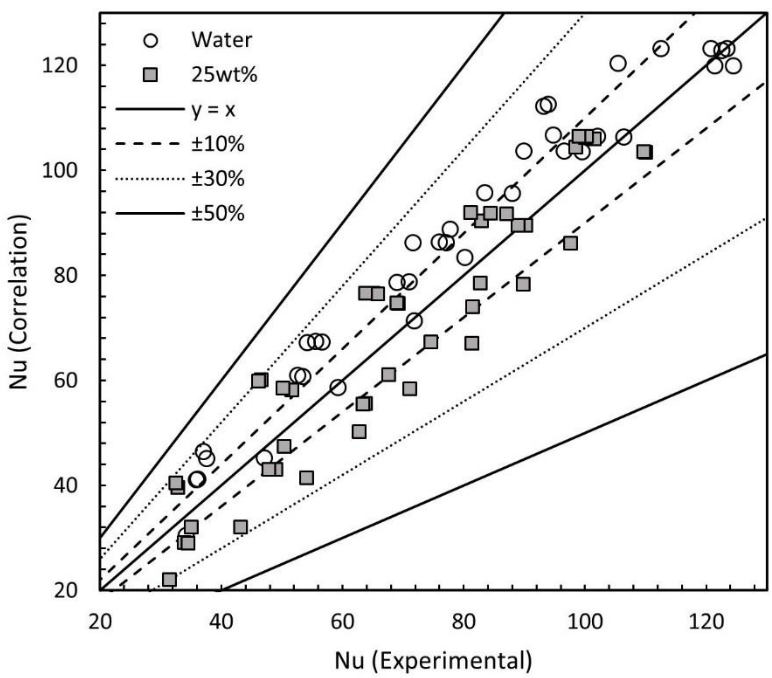

Figure 10. Comparison of Nusselt number prediction by correlation and experimental results

The correlation is further validated using data from Mackley and Stonestreet [9] as shown in Fig. 11. 72% of the data set is shown to fall within 30% confidence limits, while all of the data is within 50% limits. Only average values of Pr (73) and Re_o (300, 450, 680 and 800) could be derived from the paper. If "real" values were used, the fit would probably be better.

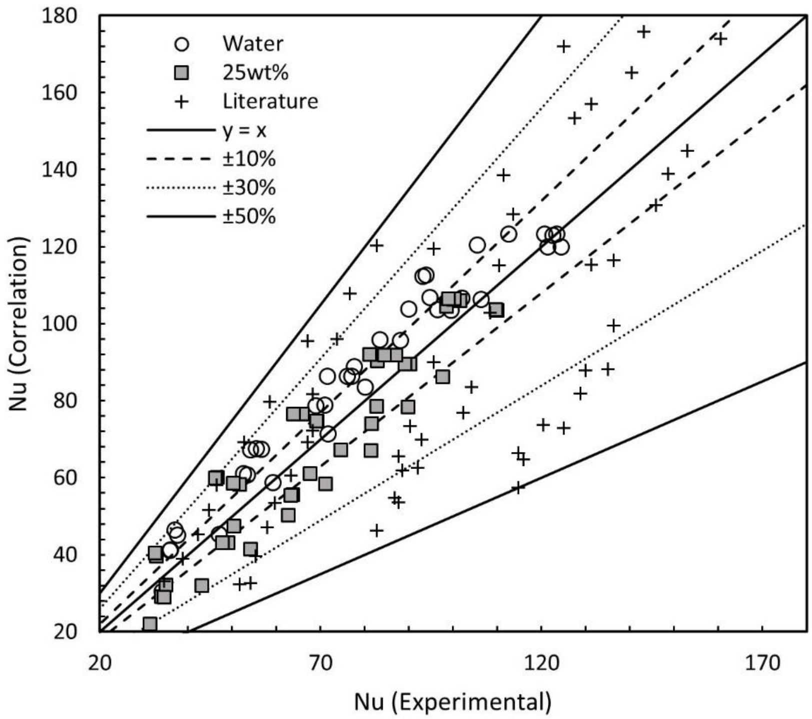

Figure 11. Comparison of Nusselt number prediction by correlation and experimental results from both this study and literature

4. Conclusions

In this paper the most comprehensive experimental study to date of heat transfer in oscillatory baffled reactors was performed. The aim was to add further depth to the understanding of the heat transfer phenomena in these novel flow-reactors, and develop new predictive correlations to aid in design of these reactors.

The maximum observed heat transfer enhancement (compared to the steady flow, unbaffled base-case) was around 5-fold while the maximum enhancement upon addition of the oscillatory flow component (compared to the steady-flow, baffled case) was around 1.7-fold.

The data show that the maximum heat transfer enhancement is observed at an oscillatory flow Reynolds number of around 1300, which is likely the point of maximum radial mixing in the OBR. This important result can be used as a guideline for any future investigation into using oscillatory baffled flows as an active enhancement technique in advanced heat exchanger design.

A new correlation for predicting film heat transfer coefficients in oscillatory baffled flows has been devised which is shown to be accurate within 30% across the experimental range investigated in this study. The correlation was further validated using data from the literature. So far it has proven to be valid for liquid flows in the range of $200 \leq Re_n \leq 1300$, $0 < Re_o \leq 8700$, $4.4 \leq Pr \leq 73$. This new correlation should improve OBR design, as it is more accurate than previous correlations and covers a significantly wider range of operating conditions, particularly with regard to Pr.

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